PATENT SPECIFICATION

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COMPLETE SPECIFICATION.

Improvements in and relating to Screw Pumps or Compressors.

We, The British Thomson-Houston Company, Limited, a British Company having its registered office at Crown House, Aldwych, London, W.C. 2 (Assignees of Allgemeine Elektricitäts-Gesellschaft, of Friedrich Karl-Ufer 2—4, Berlin, N.W., Germany, a German Company) do hereby declare the nature of this invention and in what manner the same is to be performed, to be particularly described and ascertained in and by the following statement:—

It is known that a right-handed and a left-handed screw, on parallel axes, which are in engagement one with the other and are surrounded by a casing tightly enclosing their outer circumference, can be employed as a force pump and both screw-shafts may be coupled by toothed gear-wheels to relieve the screw-threads from the transmission of the drive moment. It is further known to fit two pairs of such screws to two shafts in counter relation so as to relieve the shaft bearings from the axial pressure of the screws. Such pairs of screws can be worked as compressors or expanders of elastic fluids if their screw-threads are narrowed or widened in the course of the current.

The invention relates to improvements in such screws for the compressing or expanding of elastic fluids, for the purpose of reducing the gap-losses in the engagement areas of the screw-threads, for a favourable adaptation of the course of the thread cross-section to alterations of volume in the fluid, the coupling of more than two screws to one shaft and the formation of the screw-thread profile with a view to its easy reproduction.

As is known, screws of equal pitch and equal rectangular thread cross-section cannot be brought into engagement with each other if their axes are parallel, because their threads partially overlap, If the screws are formed with triangular or sharp trapeze-shaped threads, engagement and a lineal closure at the contacting cross-section, i.e. in their common axial central plane, can be effected, if tooth and space are made exactly alike. Outside of this plane, however, increasing gaps occur in the scope of engagement between the flanks of the threads as the profile becomes increasingly sharp, by which gaps spaces of different pressure are connected and consequently gap-losses originate.

Gaps of this kind are hereinafter termed "flank-gaps" in contradistinction to the "profile-gaps" which can occur in the plane of contact of the two screws. In this invention only gaps and gap-losses are considered which are due to the formation of the screws, and not those between the circumference of the screw and the casing, or those due to considerations of friction, inaccuracies in production and differences in heat expansion.

As will be hereinafter shown, it is not possible to simultaneously avoid both profile and flank gaps; the invention however affords, with geometrical similarity of the successive threads, means for restricting to the smallest measure the remaining gaps.

If the generatrix of a screw-thread surface is a straight line which stands perpendicularly to the screw axis, then an overlapping occurs between the flanks of a right- and a left-hand screw brought into engagement, as above, which overlapping can be removed if the thread-spaces are made broader than the profile of the thread, or if a somewhat involute like taper is imparted to the latter at the outside. In both cases both profile and flank gaps result, which are however admissible in so far as the screws serve for the transport of non-elastic liquids, but not for elastic liquids, in connection with which even slight gap-pressure differences generate extraordinarily high current speeds in the gaps. The gap losses of the elastic working medium can be suppressed by an addition of small amounts of a non-elastic liquid, but as driving power has to be supplied for the non-elastic packing material also, which must be taken into consideration in the total balance of the machine, the gaps must still be kept as small as possible.

Figs. 1—6 of the accompanying drawing illustrate the gap conditions possible between two screws with right- and left-hand threads. If Fig. 1 is the axial

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projection for two neighbouring screw-flanks, with straight rectangular generatrices having r_{1i} and r_{2i} as inner radii and r_{1a} and r_{2a} as outer radii, then gaps result before the centre of the axis $o_1—o_2$, which gaps increase from the central tangent towards the left, while overlappings increase towards the right, and between BC and AE line-contact takes place. The size of the respective overlapping of the gaps and their course over the distance EA is indicated by a series of cross sections designated by U and S; and its maximum value, extending along the outside edge of a thread, is shown by the lateral projections of Fig. 2. If the outer circles of the spiral surfaces are allowed to stand, then a rectangular thread-profile must be undercut in known manner, as is represented in Fig. 3. The undercutting is greatest in the foot of the profile, and increases if the exterior diameter remains the same with decreasing diameter of the core. Spirals produced with profiles of this kind result (in the scope of engagement) in lineal closure between AB and CE. The edges AC and BE remain open and provide for the through-passage through the profile gap, which runs from the bottom of the thread in O_2 to the communicating one in O_1 in the direction of the arrows $S_2—S_1$.

The production of spiral passages profiled in this way is possible for screw-pumps; the thread having a constant pitch and depth and being produced on the lathe. If, however, compressor-screws are to be produced on the lathe, this is only possible by varying the depth of the thread; since the lathe only supplies a constant pitch (See Fig. 7). In this case the ideal minimum gap varies, according to Fig. 3, over the whole course of the thread both in the length and breadth and is consequently not produceable on the lathe. It is possible to make compressor screws having a cylindrical external diameter and also cylindrical diameter at the bottom of the thread if their pitch varies, as shown in Fig. 4. Such screws would be suitable as regards the depth of thread for the application of the form of tooth, Fig. 3, but not as regards the pitch. This necessitates an undercutting in accordance with the greatest pitch, and would have to be retained for the whole screw. By this, the gap-values for the smallest pitch would become inadmissibly great and the closing-up at the flanks would be lost.

This overlapping of the screw-flanks can be avoided as already mentioned by the employment of a trapeze-shaped thread, if the acute angle of the trapeze is sufficiently great so that the inner edge of a thread-gap drops towards the outside, owing to the conical form, far enough to prevent the outer circle of the counter-screw from overlapping it. In the ideal case, this condition cannot be practically fulfilled, if the trapeze inclination forms a tangent at the gap-curve of the straight flank. In Fig. 3 such an inclination is indicated in dotted lines. A trapeze form of this kind gives a perfect lineal closure in the profile of contact of the longitudinal plane $O_1—O_2$, but it results in flank-gaps transversely to AE, which again have their minimum values between AB and CE, and the form shown in Fig. 5 with the maximum axial dimension f_1. Since they occur doubly in each flank engagement and are longer than the profile-gaps, they will always result in greater values than these.

The size of the gap cross section is however not alone the measure for judging of the gap losses, but only in connection with the pressure-differences between the threads and between the pressure-grades between which the gap lies. In this respect, profile-gaps are substantially more unfavourable than flank gaps, as will be seen from Fig. 7. Here two screws are illustrated with four threads engaging in each other, the closed pressure-steps of which, i.e. those which show a difference as against the ingoing and outgoing pressure, are numbered 1 to 4. The step indicated at 0, which has just closed, still possesses the entrance-pressure; the last in the succession, indicated at 5, must already have the out-going pressure, since it stands immediately before opening. As will be seen by following the dotted lines indicating the bottom of the thread-spaces, the pressure-steps numbered in brackets connect with the pressure steps correspondingly numbered, beyond the central plane. From this it follows that of the two sequencies of five steps, each having one pressure grade difference, those inscribed to the left of the figure have leakage passages produced by the flank-gaps, those inscribed at the right hand side, by the profile gaps. Thus the smaller profile gap cross-section is here equalised by greater pressure-differences (more or less pressure) and it is just as important to make the profile as the flank gaps small.

Thus we find :—(1), that perfect flank-closure, which should extend in the line of the central plane AE of Fig. 1, is not attainable on account of the overlapping; (2), perfect profile closure is inadmissible because it results in too great flank-gaps; (3), curved forms of the thread flank profile are not feasible with geometric similarity with the tools available, if the cross-section of the thread varies permanently

along its path. The object of the invention is the production of compressor screws, the profile-flank lines of which are only straight geometrically-similar producable forms, and the dimensioning of these in order to attain simultaneously small profile and flank gaps. This is attained according to the invention, by the employment of a trapeze shape forming the thread-space, in connection with which the generatrix of the thread-flanks only deviates from a rectangular generatrix to the extent of a very acute angle, in the order of magnitude of about 3°. In this connection, in as far as an overlapping remainder still remains at the bottom of every flank, this can be removed in a second operation, in that the spaces between the teeth are widened in the outer part by a second trapeze form having a somewhat greater flank angle, as shown in Fig. 6. In Figure 6a, the form of the space is shown in detail and, for the sake of clearness, the size of the flank angle is somewhat exaggerated. With the space and thread-profile illustrated in Fig. 6, the maximum value of the flank-gap is reduced from f_1 to f_2 as will be seen from Fig. 5. According to the selection of the angles, the point of intersection of the two flank angles may be displaced between the centre and outside edge of the profile, so that small triangular profile gaps form at the foot and head, and the flank-gap is simultaneously reduced to as small a measure as possible. The profile gap according to Fig. 6 experiences at both sides a further throttling in that, in the course of the line D F (Fig. 1) flank-closure occurs, in which connection flank-gaps result between DB and CF which open the way to CB, but are smaller than the profile-gaps.

Fig. 8 shows the section through a pair of screws according to the invention, in which the economical form for a given number of steps is attained, having regard to the constructional length and gap losses, when the depth of space h is equal to the corresponding half-pitch, so that the mean space-breadth b at each point of the thread is approximately equal to h. The geometrical similarity thus pronounced of all the threads is not only a condition for $b/h \sim 1$, but also for all other values, through the possibility of production which only admits of the execution of straight cones for core and external circumference of the screws with an angle of enlargement d, the mean value of which coincides with the enveloping line of the mean screw-cylinder. In order better to adapt the enlarging or tapering ratio of the screws, which determines the magnitude of d, to the course of the specific volume and equal capacity distribution over the axial spiral-path, the screws are composed of two or more parts having different angles of enlargement, as for instance T_1, T_2, T_3 in Fig. 9. Each of these parts is then machined independently on a screw-cutting machine. Advantageously then the geometrical progression of the b-values of T_1 is continued for T_2 and T_3 and only the depth of h is adjusted for another d, because in this case the end-section surface of the outgoing and in-going threads fit accurately together. Screws of the above described kind can be employed either pair-wise or, for greater capacities, two or more can be distributed around a central screw and can be coupled thereto by gear wheels.

The production of the screws can take place in this way, that patterns are made in the form of the screw with allowance for machining, so that castings made therefrom only need a slight turning down to finish them. The finishing can take place in this way, that a right- and left-hand screw is made as a cutting tool by the aid of which the previously cast screws can be finished in that the piece of work and the tool are driven at the same angular velocity and are brought into engagement by slowly reducing the distance between their parallel axes.

In order to reduce an undesired heat flux with the screws in the axial direction, the screws can be made of material having substantially inferior heat-conductivity to that of metals, for instance substances containing the products of polymerised synthetic resins. In such case the formation of the screws can take place not only by cutting, but by pressing.

In order to improve the packing of the screws against the casing, they can be provided at their outer circumference with outwardly spring-yielding packing rings, after the manner of piston rings, which vary in cross-section and in their thread ratios according to the screw threads and are subdivided for each thread-winding into single members which may embrace an angle of 360°.

Having now particularly described and ascertained the nature of our said invention and in what manner the same is to be performed, we declare that what we claim is:—

1. A screw-compressor comprising right- and left-handed screws coupled in engagement with each other and closely enclosed in a casing, characterised in that the space between the teeth is formed by a basic-trapeze of very acute angle and a trapeze of a somewhat greater angle overlapping the same.

2. A compressor screw according to

Claim 1, characterised in that the depth of the screw thread or the pitch or both, vary according to a geometrical progression of the form $a, aq, aq^2 \ldots aq^{n-1}$.

3. A compressor screw according to Claim 1, characterised in that the depth of the screw-thread is equal at every point to half the thread-pitch belonging to it.

4. A screw-compressor according to Claim 1, characterised in that with a central screw two or more screws of reversed thread and of the same or half the diameter are in engagement and are coupled by gearing, the outer screws, if of half the diameter, having double the speed of the central one.

5. A compressor screw according to any preceding Claim, characterised in that it is cast in the rough and only finished in a shaping machine.

6. A compressor screw according to any preceding Claim, characterised in that it consists of a material having poor heat conductivity.

7. A compressor screw according to any of the Claims 1 to 4, characterised in that its exact form is produced by a pressing operation.

8. A compressor screw according to any of the Claims 1 to 4, characterised by a groove on the outer circumference extending in the form of a logarithmic spiral and a spiral ring within the same of like form, having an outward spring, after the manner of piston rings, and which is sub-divided into single members embracing 360°.

Dated this 3rd day of January, 1934.

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Agent for the Applicants.

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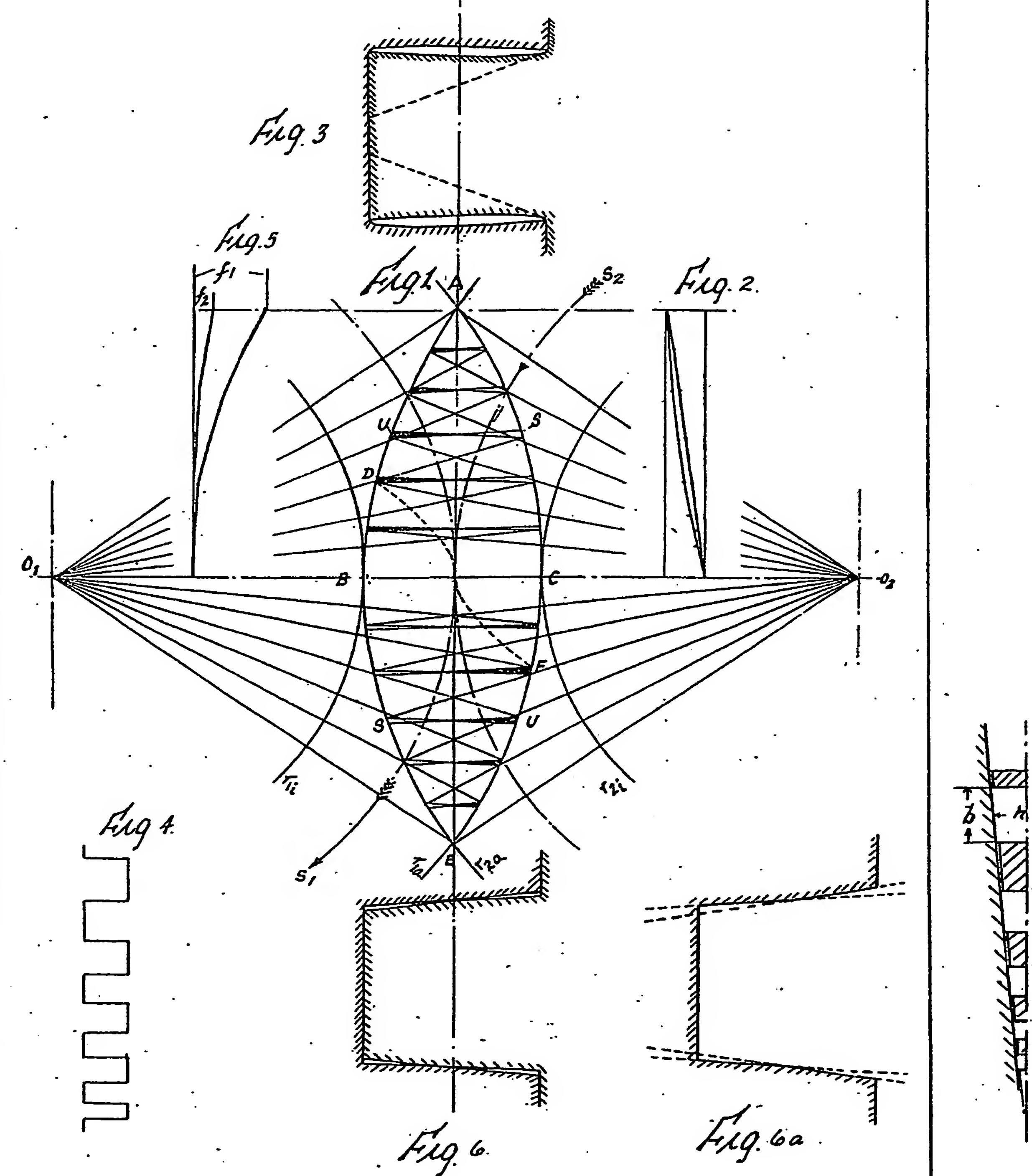
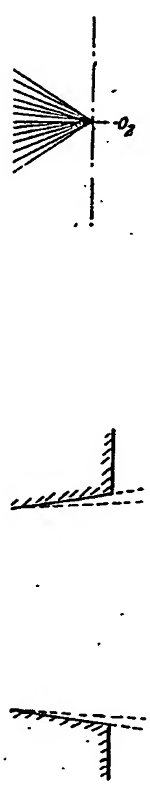
SHEET 1

[This Drawing is a reproduction of the Original on a reduced scale.]



SHEET 1



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SHEET 2

Fig. 7

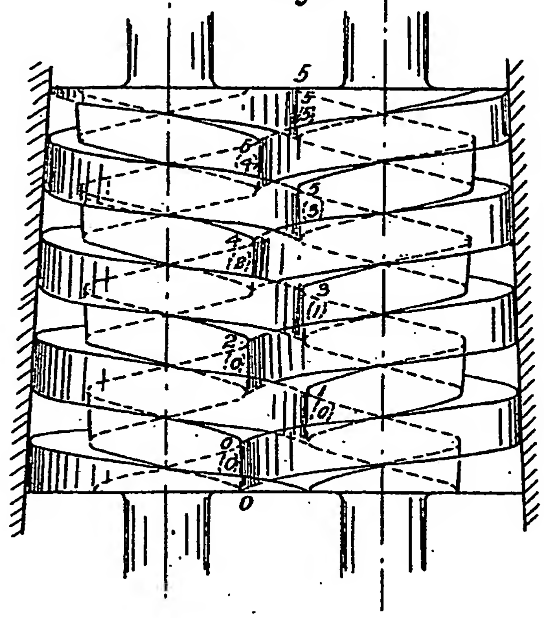


Fig. 8

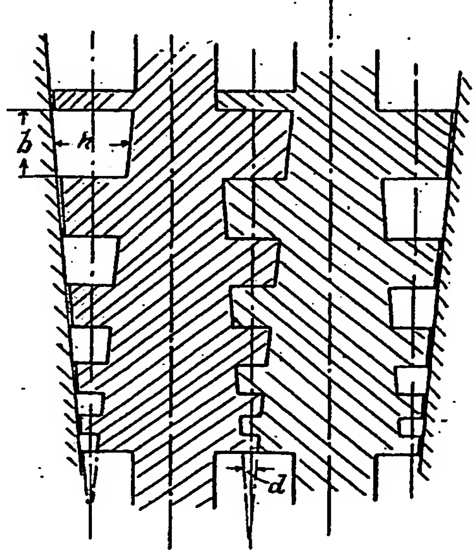
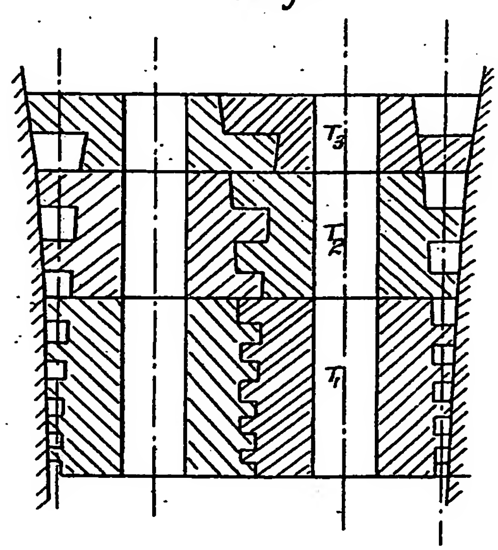


Fig. 9



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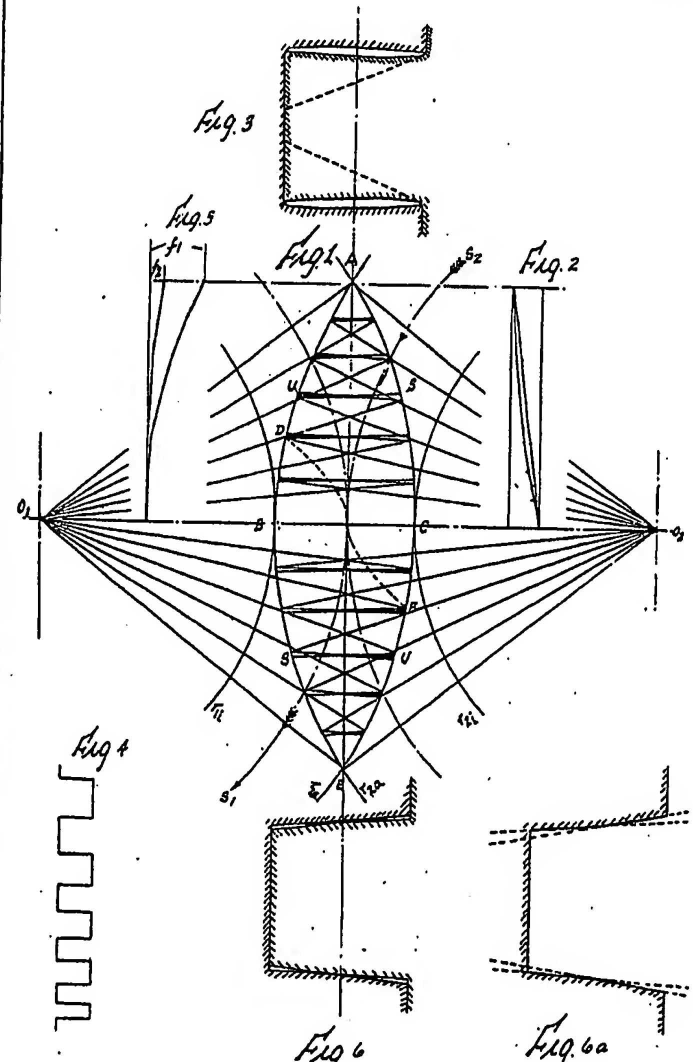

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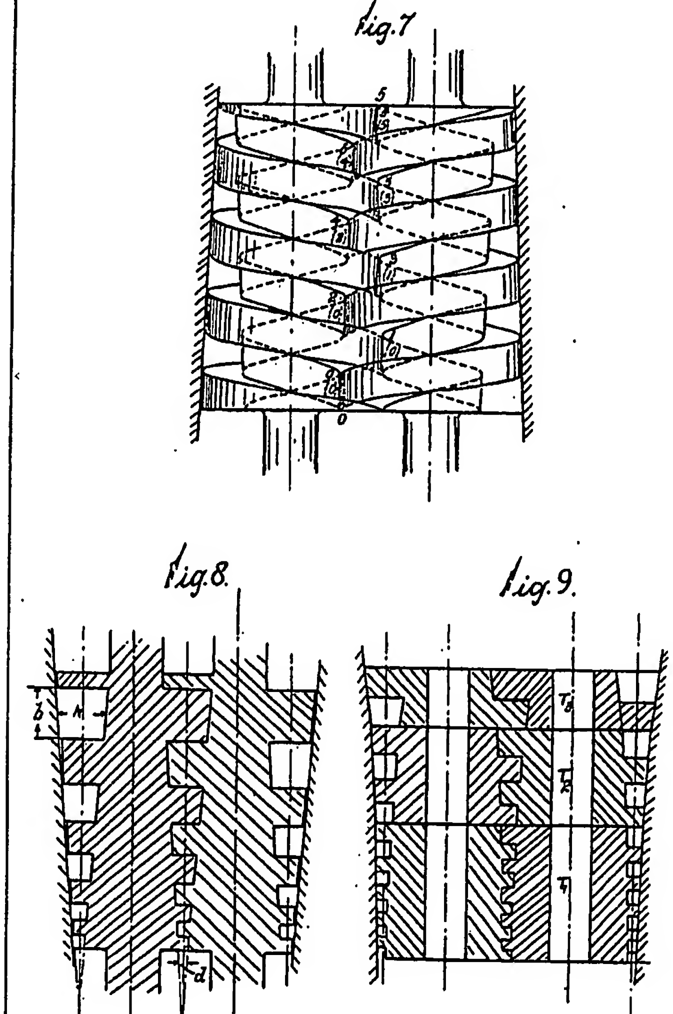

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